NEW DEVELOPMENTS IN CENTRIFUGAL COMPRESSORS

by
John D. Halloran
Manager of Engineering
Joy Manufacturing Company
Buffalo, New York

John D. Halloran is Manager of Engineering for the Turbocompressor Division of Joy Manufacturing Company. His responsibilities include management of the engineering and field service operations for this manufacturer of integrally geared centrifugal compressors. He holds a B.S. degree in Mechanical Engineering from the University of Illinois and a M.B.A. degree from the State University of New York. He is also a member of ASME.

ABSTRACT

Design goals and operating experience with a new bearing design and a new microprocessor based compressor control system are discussed. Both of these developments are included in a compressor designed for the small centrifugal plant air market, but potential applications exist beyond this single type of turbo machine.

The new bearing design is discussed in terms of its usage with two gear driven pinion shafts, one running at 36,000 rpm and the other running at 50,000 rpm. Major design aims achieved were for increased effective rotor/bearing system damping, rugged construction features and ease of assembly and maintenance.

The microprocessor based compressor control system includes all normal monitoring functions plus adaptive digital control techniques with memory based learning. The dual loop interactive system is equipped only with setpoint adjustments, as the control is capable of self-adapting its PID values based on the last incremental system response.

INTRODUCTION

This paper discusses two new design developments that have been included in a small packaged centrifugal plant air compressor, which is shown in the photograph of Figure 1. The compressor covers the range of 300 horsepower to 500 horsepower, and is an integrally geared design with impellers for the individual stages of compression mounted on the pinion shafts. The low speed pinion is a single overhung design operating at 36,000 rpm, and the high speed pinion is a double overhung design operating at 50,000 rpm. The compressor is controlled by modulating inlet guide vanes which may operate in any position from the fully closed (no air demand) to fully opened, depending on ambient conditions and level of demand on the air system being supplied.

First, a new bearing design is discussed and operating experience presented, and, second, a compressor control system is presented which includes the use of current electronic technologies and new philosophies of operation. Both features are considered to be departures from traditional design prac-

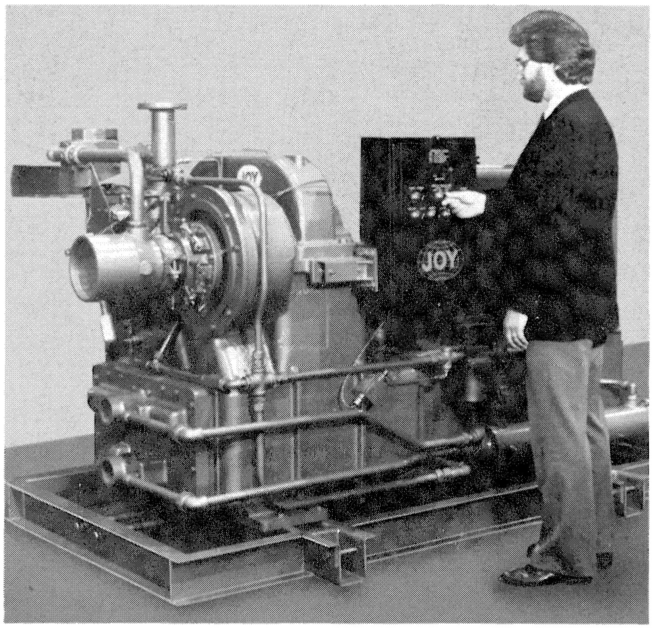

Figure 1. Integrally Geared Centrifugal Compressor Package.

tices, and from that stance merit discussion regarding the advantages which they offer and the potential for use in other applications of rotating machinery.

The two subjects of bearings and controls are diverse in the technologies they represent, and the vocabularies used for description. Including both subjects in one paper of limited theoretical depth is premeditated to emphasize the point that rotating machinery professionals should be generally conversant in several areas of technology. When the subjects are as diverse as rotor/bearing dynamics and microprocessor based closed loop control systems, the required knowledge may seem extremely broad but the single subject intended is operation of one total machine package.

Examples of the need for emphasis on total machine operation can be sighted where mismatches have occurred between controls designed by one company and machines designed by another company. Better understanding and coordination of overall machine operation would have prevented those problems. Said more strongly, new control technologies should not be totally vested in a separate group of instrument experts and computer programmers. There should be a constant effort to force information exchange across any such interface.

HYDROSTATIC SQUEEZE FILM BEARING

The concept of designing rotating machinery with viscous squeeze films that are concentric to the machine's radial journal bearings has been discussed and written about for a number of years. Much analytical modeling effort has reached the available literature [1] through [5] and various mechanical designs have been conceived [6] through [10]. All of this work is motivated toward providing more energy-absorbing damping into rotor/bearing systems. This damping of rotor/bearing systems is desired to suppress the amplification factors of system resonances and to counter sub-synchronous whirl phenomenon which can be caused by a multitude of forcing systems.

In general, the world seems to provide excitation for rotor/bearing vibration. Efforts to define and minimize these excitations are necessary, but so are the efforts to design machinery with higher levels of effective rotor/bearing system damping to absorb vibrational energy. Interestingly, it can be shown analytically that rotor/bearing systems can be overdamped, but this author has not knowingly seen such a system in hardware. The normal turbomachine could always use more damping.

When used correctly, viscous squeeze films add a new dimension to machinery design. They allow the design of the load carrying hydrodynamic bearing with major emphasis on the concerns of optimizing the static and dynamic *load* carrying capability under design imposed spectra of speed, load and viscosity changes. The squeeze film design then carries the emphasis for optimizing the total rotor/bearing system effective *damping* under the conditions of varying load and viscosity, but at zero speed. Compared to the balancing act of achieving high stability margins and conservative load carrying capability in only one hydrodynamic oil film, the additional viscous squeeze film allows considerably more design flexibility.

In an integrally geared compressor, the operating demands on the bearing system are broad because of the wide variations of load, viscosity and attitude angle. Gear reaction load is the main part of the total bearing load when the machine is operating at design horsepower with pinion weight contributing only a minor amount. When the compressor is fully throttled, and gear reaction load is low, pinion weight may become dominant. Under this condition, the attitude angle may shift as much as 180° on pinions that are being driven upward. Other operating necessities include viscosity changes between hot and cold oil and the dimensional changes caused by thermal effects during the transients of fast motor starts. In addition, consideration that most pinions run at more than twice their first peak response frequency gives appreciation of the potential usefulness of viscous squeeze films in internally geared centrifugal compressors.

The hydrostatic squeeze film bearing shown schematically in Figure 2 was developed with the belief that it offered the many advantages of viscous squeeze films, along with a number of additional gains when compared to other designs proposed in the literature. The distinct advantages of this bearing which were foreseen in the design stages, and since confirmed in hardware, are as follows:

1. The use of hydrostatic pressure to automatically center the journal bearing within the squeeze film circumvents problems associated with mechanical centering devices such as assembly setups, predictability and repeatability of radial spring rates, and/or concentrating dynamic radial bearing loads to localized wear points. This last point is quite important to long-term operation. In the hydrostatic squeeze film bearing, all radial loads are carried only by oil films over large areas, providing a very rugged construction to withstand the dynamic loads of turbo machinery.
2. The use of relatively high pressure to provide the hydrostatic action in the squeeze film provides protection against squeeze film cavitation under all plausible conditions of vibration amplitude and frequency. Cavitation in the squeeze film is to be avoided, since it causes major decreases in available damping and "pseudo" spring rates that may allow unstable "jump" actions in squeeze film bearing operation [11].
3. Design flexibilities to allow damper eccentricities to be kept low and the inner hydrodynamic film relatively stiff compared to the squeeze film so that bearing characteristics change only slightly with changes in bearing load, attitude angle of the bearing load, and lubricant viscosity [12].
4. Three dimensional self-alignment of the bearing assembly in the squeeze film clearance.
5. No additional oil flow requirements over that required by the hydrodynamic bearing, since the supplied oil is first used in the squeeze film before passing to the hydrodynamic oil film.
6. The hydrostatic squeeze film configuration requires only conventional machining capabilities and standard tolerance levels for bearings. It also allows a horizontally split design for ease of assembly and inspection.

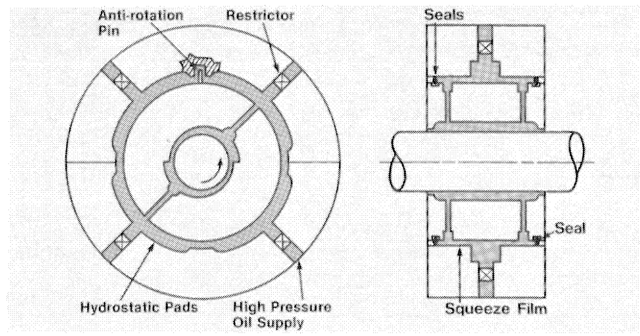

Figure 2. Schematic Arrangement of a Hydrostatic Viscous Squeeze-Film Bearing.

The specific design of hydrostatic squeeze film bearing being used in the small centrifugal compressor is discussed more fully in [12], and is shown in the photograph of Figure 3.

The curve shown in Figure 4 is taken from [12], and represents an analytical study of the stability of the double overhung 50,000 rpm pinion operating on offset half bearings, idealized (no cross coupling) 5-pad-tilting-pad journal bearings and offset half bearings in conjunction with hydrostatic squeeze films. Figure 4 does take some explanation. Its basis is a series of analytical stability studies of the loaded compressor varying only bearing clearance or only damper clearance in the case of the hydrostatic squeeze film application. The lowest log decrement of all of the peak response frequencies predicted for each stability run is then used as one point on each curve.

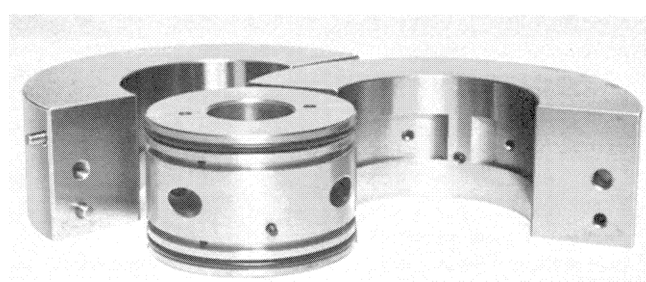

Figure 3. Hydrostatic Squeeze-Film Bearing.

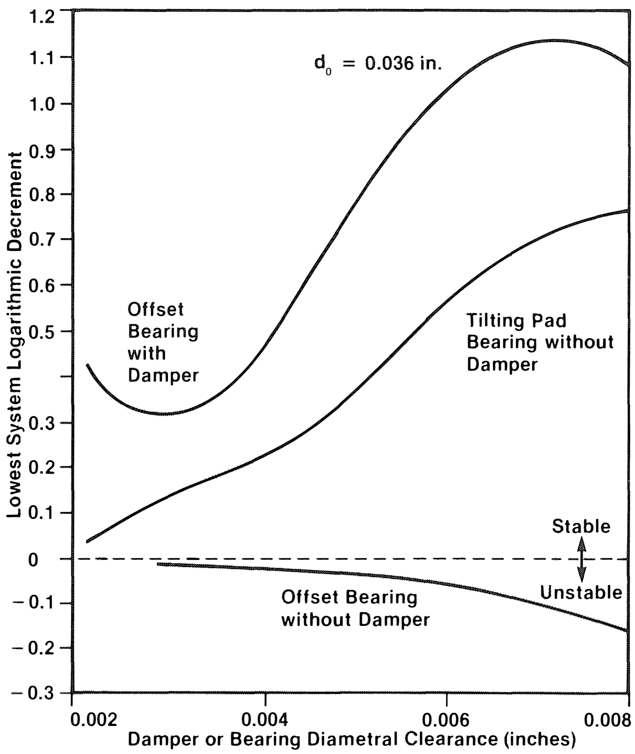

Figure 4. Loaded Compressor Rotor/Bearing Stability Plot. d_o = orifice diameter.

The plot of lowest logarithmic decrement in Figure 4 for the offset bearing shows that operation is predicted to be unstable for all levels of bearing clearance. The offset bearing was chosen for comparison because, under the given conditions, it is predicted to be the most stable fixed pad bearing design of the several analyzed by [13] and [14]. The plot of lowest logarithmic decrement for the 5-pad-tilting-pad bearing shows good stability margins with 0.006 to 0.008 inch diametral clearance on this 1.25 inch bearing. The stability plot of the offset bearing with the hydrostatic squeeze film shows even more stability margin than the tilting pad bearing. With a squeeze film clearance of .007 inch diametral, the stability margin is nearly 60% better for the hydrostatic squeeze film bearing when compared to the idealized 5-pad-tilting-pad bearing.

It needs to be emphasized that the hydrostatic squeeze film bearing is being presented as an additional alternative for the turbomachinery designer. Many factors must be evaluated in choosing a bearing design for a particular application. The data of Figure 4 shows stability margins for one particular application.

The history of the hydrostatic squeeze film bearing includes 3 years of in-house development, followed by nearly 2 years of closely monitored field testing. The field testing was done on an operating compressor equipped with four of the new bearings, two on each pinion. This prototype compressor was also equipped for remote data retrieval, and Figure 5 shows vibration data collected at regular intervals by phone over the last 21 months of the field test period. The data of Figure 5 is in the form of the first three harmonics of pinion vibration and shows no major change from the "as shipped" amplitudes which are also indicated on Figure 5. Throughout this entire field test period, no significant sub-synchronous vibration amplitudes were measured. At the end of the two-year test run, the compressor was upgraded in peripheral equipment to the new production configuration, and is now back in operation with its original bearings.

Further experience with the hydrostatic squeeze film bearing design has been gained at a fast pace as production compressors with the bearing have been built, tested and put into field operation. A measure of successful rotor/bearing performance can be gained from a review of the radial proximity probe vibration readouts that are taken from the pinion shaft as close to the overhung impellers as is practical. A survey of the first 30 production machines, running full speed and full load on the test stand, with three vibration readings per machine (90 vibration amplitudes) showed a mean unfiltered amplitude of slightly less than 0.3 mil peak-to-peak with a standard deviation of less than 0.09 mil. Those values are down in the noise level of unfiltered proximity probe data, and certainly verify that the hydrostatic squeeze film bearings are performing as planned.

Field startup of the compressors has been equally gratifying. Although the myriad of startup mishaps which can cause unexpected strains on machinery have occurred, not one bearing or rotor malfunction has been reported. That record has maintained as field population of the hydrostatic squeeze film bearing as well as operating history has increased at a rapid rate.

CONCLUSIONS (HYDROSTATIC SQUEEZE FILM)

The general conclusions that can be reached from the experience to date regarding the hydrostatic squeeze film bearing are as follows:

1. The initial application of the hydrostatic squeeze film bearing to an integrally geared centrifugal air compressor is judged to be highly successful based on extended field usage in a significant number of different compressor applications.
2. The hydrostatic squeeze film bearing can be considered as another alternative to the turbo machinery designer when the desire exists for additional rotor/bearing system damping above that available from the various other popularized bearing designs.
3. The only disadvantage of the hydrostatic squeeze film bearing defined to this point in time is in the extra horsepower required to provide oil pressures to hydrostatic levels. In the case of the present application to an integrally geared centrifugal air compressor, this power penalty is on the order of 3 horsepower, which is 1% or less of overall compressor power. That power requirement is somewhat offset by the reduced oil flow required when compared to a tilting pad bearing designed for the same duty.

ADAPTIVE COMPRESSOR CONTROL

The general microprocessor based revolution occurring in the world today is being caused by the speed and power of these marvelous electronic devices which are now available at very low hardware costs. It is believed that these electronic technologies should bring a new era of intelligent machinery, whose inherent distributive knowledge to monitor, diagnose, communicate and control is only limited by our present-day imagination. That statement is not meant to convey that the limitation is a minor one, because complex software creation is time-consuming and the source of the real costs. But without

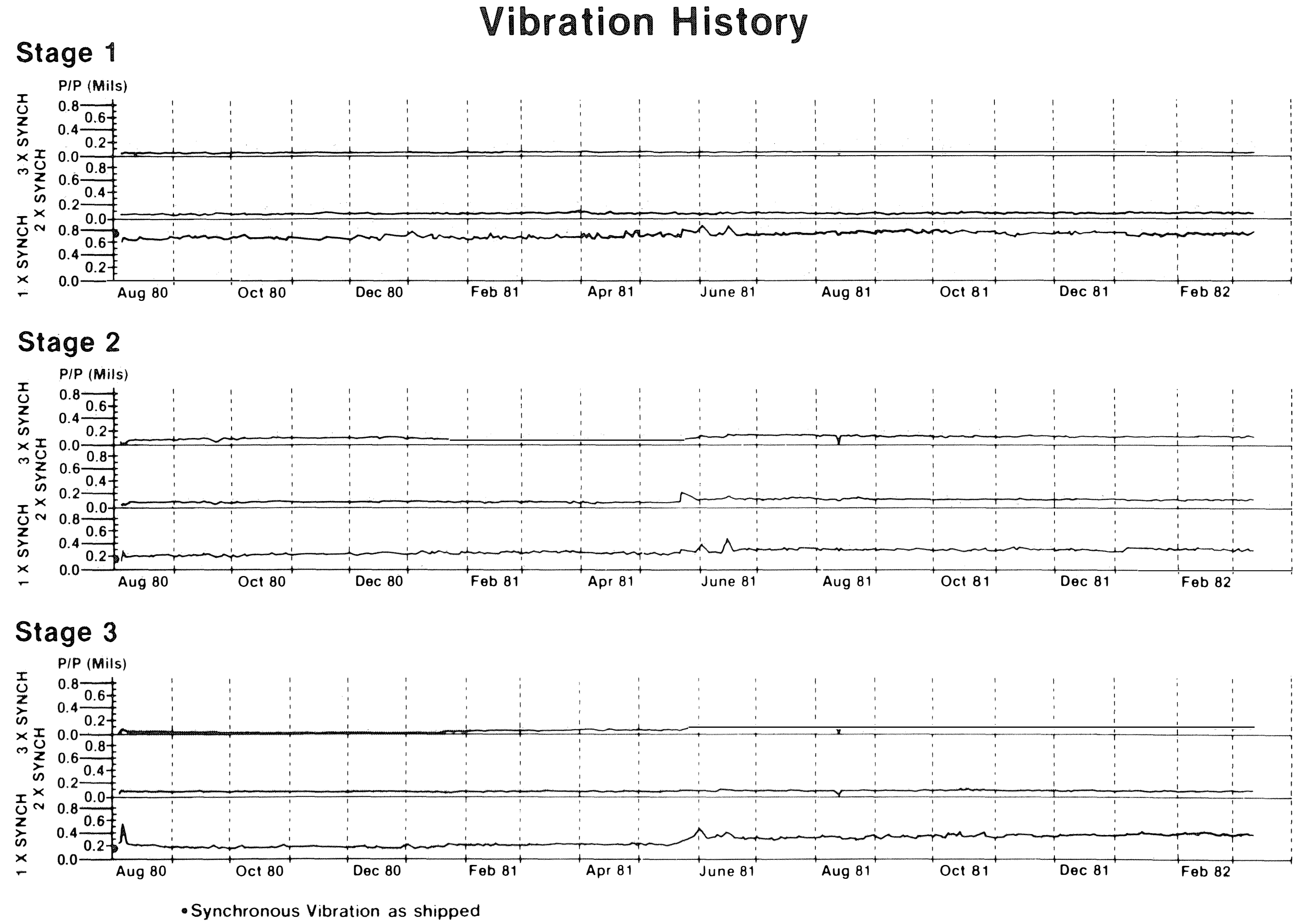

Figure 5. Vibration History Plot of Compressor Field Test using Hydrostatic Squeeze-Film Bearings.

the up-front imagination, the targets will be set far under the capabilities of the electronics.

As a demonstration of the knowingly gross statements expressed in the preceeding paragraph, this section of the paper is intended as a descriptive report of experience on an initial effort at incorporating a microprocessor based monitoring and control system on a centrifugal compressor. Interestingly, this first generation electronic control is incorporated on the same small integrally geared centrifugal compressor that was described in the first half of this report as being equipped with hydrostatic squeeze film bearings.

A major point to be stressed in the following description is that the total control function accomplished could not be done practically with pneumatic and/or analog systems. This means the design aims were beyond present-day practices, and still were accomplished without using most of the microprocessor's capabilities. The design aims were high, but could have been a lot higher.

A number of open loop monitoring functions are incorporated in the subject compressor control system. The microprocessor software is interrupt structured to monitor oil temperature and pressures and air temperatures, and will provide safety shutdowns with first-out display when abnormal conditions are defined. The system also has three vibration measuring proximity transducers to read vibration levels from the pinion shafts near the overhung impellers. The analog output of the vibration transducers is digitized at a rate of over 5,000 Hertz, and a true peak-to-peak amplitude value for the measurement period is retained in a first-in, first-out averaging buffer memory. The averaging technique is used to give the desired integrated time delay to each vibration signal before comparing it to alarm and shutdown setpoints. Also, the vibration transducer output DC voltage is monitored by the microprocessor-based digital system and will cause an alarm condition if the transducer or its supply voltage fails, or if the probe gap is outside the linear range of the transducer system. All of the above monitoring functions are considered standard for a small centrifugal compressor and are, in this case, accomplished with digital logic rather than analog circuitry.

The overall compressor control function is as illustrated in Figure 6. In the loaded operating condition, the compressor bypass valve is closed and inlet guide vanes mounted in front of the first stage of compression are modulated to control to either a maximum motor amp setpoint (MAXAMP) or a discharge pressure setpoint (SETPR), according to which variable is in control. If the compressed air demand drops off and the control closes the inlet guide vanes until the amps reach a minimum setpoint (MINAMP) level, the compressor unloads to its fully throttled condition by closing the inlet guide vanes and opening the bypass valve. When the system pressure falls below the setpoint pressure by specified amount, the compressor automatically reloads. The specified amount of pressure drop is the fourth adjustable setpoint (DP) in the control system.

The above description, along with Figure 6, shows that

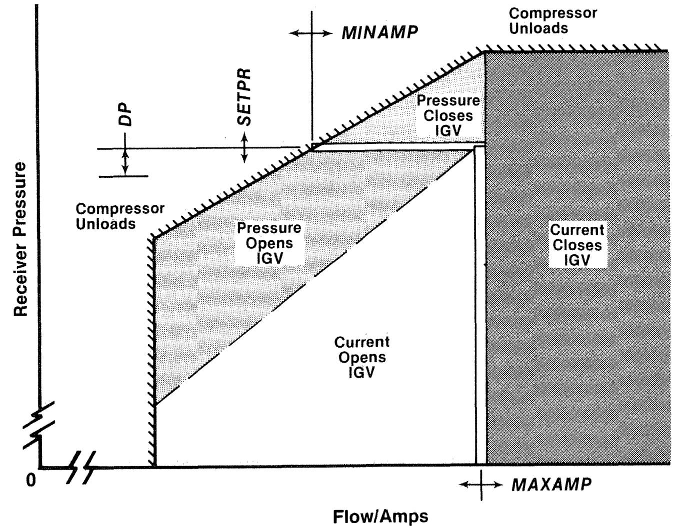

Figure 6. Compressor Control Map.

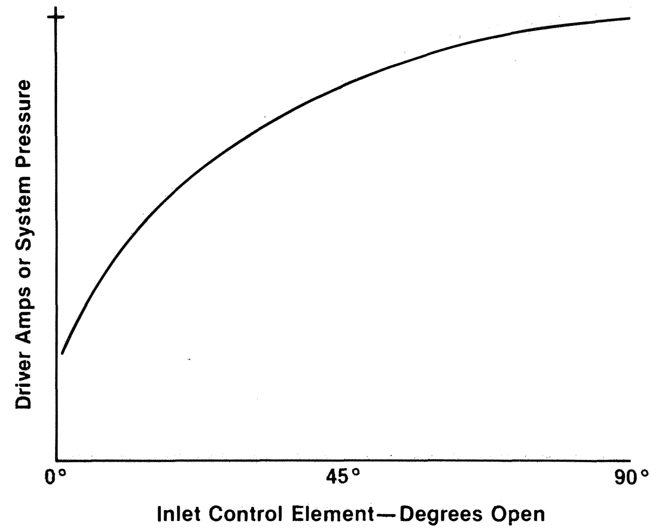

Figure 7. Schematic Representation of a Non-Linear Compressor Control Characteristic.

the compressor control has two closed loops of control to setpoint levels of motor amps and system discharge pressure. The control additionally has two setpoints, minimum amps and system pressure drop, that act as trigger points. The two closed control loops are of special interest because they have been made adaptive; that is, the loop adjustments for proportional band (gain) integral (reset) and derivative actions are internal to the digital control logic, and only setpoint adjustments are made to the control.

A review of literature in regard to adaptive control can be summarized by saying that adaptive control logic is spoken of fondly, because it allows efficient (tight) control over wide ranges with highly non-linear elements within the control loop and with major fluctuations of the controlled variable. These characteristics of a control loop fit the plant air compressor application quite well. Adaptive control does not, however, fit nice mathematical formulation for computerized evaluations of stability. As the literature speaks with praise, it suggests other methods for approximating the capabilities of adaptive control. Also, control literature mentions the problems of signal noise as being especially bad in terms of confusing an adaptive control and causing instability.

In order to achieve the desired results without the problems mentioned in literature, the subject adaptive control was designed and made quite functional by skipping the mathematical modeling and by using averaging techniques on the data taken from the controlled variables so that noise effects (such as slip frequency amp excursions) were nullified. Instabilities have not been a problem, and tight control in both the amp and pressure loops has been accomplished.

Figure 7 is a schematic representation of the non-linearity present in a control element such as an inlet butterfly valve or inlet guide vanes. No vertical scale is shown on Figure 7, as the curve varies in position with ambient conditions and control configuration. The entire non-linear characteristic of Figure 7 is necessarily used on centrifugal compressors, since on cold days the inlet device will operate very near the closed position and on hot days it will operate near the open position. Without adaptive control, if the controller gain of the closed loop control system is set for tight control on a hot day, it will be unstable on a cold day when the control element increases the loop gain significantly. Conversely, if the controller gain is set for cold day stable operation, the hot day control performance will be inefficient (loose control) as the control element decreases the loop gain.

In the subject control system, the control logic is equipped with sufficient intelligence to evaluate the last incremental control move. It remembers the value of the controlled variable prior to the last move, knows the incremental size of the control move and learns the new end point of the controlled variable. It then uses this data to calculate the incremental gain of the entire control loop, which is the slope of the curve in Figure 7 at the existing point of control. That knowledge, along with the error signal, is used to calculate the magnitude of the next control move. In this way, the gain of the control loops are adaptive and give efficient stable control at all positions of the control element. The limits imposed within the software restrict the ratio of change in gain from maximum to minimum to a value of 4.3 to 1. This gain restriction was first tried as a matter of software convenience and has not subsequently been altered, as both control loops seem to work quite well with that level of adaptive gain change. At the highest level of gain, resolution is 0.75% in the amp control loop and 0.38% in the pressure control loop.

The control timing of the amp control loop (dead times and time rate of changes of controlled variable) is entirely contained within the supplied hardware of the plant air package, and is repetitive between compressor packages. These values have been measured and the reset time of the amp control loop set accordingly in software as a fixed value. In like manner, the measured timing values show no derivative action is required in this amp control loop since the controlled variable reacts quickly to changes of the control element.

The pressure control loop presents quite a different control challenge. In this loop the size of the supplied air system determines timing and each installation is different, plus may change over time. Also, pressure is usually a poor indicator of the magnitude of the compressed air demand change since the supplied systems have large volumes (high inertia), causing the pressure reading to react very slowly. Derivative action is desired in the control to catch major demand swings with as little overshoot as possible.

Again, in the pressure control loop the memory capability of a microprocessor system is used. In this case, the control monitors the time rate of change of the controlled variable after a control move. It then calculates a time span to the next allowed control action. Of course, if the pressure error signal continues to increase after a control move, the system makes a new corrective action quickly. The pressure control timing, which provides integral action, is allowed to vary between restraints of one second to forty-five seconds, which has proven to be quite adequate.

Derivation action in the pressure control loop is simulated by doubling the control output if the error signal exceeds a preset value. Said another way, if the system pressure gets to a worrisome level the controller output is not only proportional to the error signal but is additionally multiplied by two to allow a bigger control move to overcome the system inertia. Only one trigger level for this derivation action is incorporated in the subject software logic.

The original debugging of the control hardware, software and total compressor system was done on a plant air compressor package installed at the manufacturer's plant. This development process occurred over many months and included a number of evolutionary upgrades of the design aims as the power available in the microprocessor based electronics was realized. The first production designs of the control system contained the completely debugged adaptive logic previously described.

An additional design aim of the control development program was to take precautions to assure that the control would operate satisfactorily in industrial environments. Solid state electronics and computers in general have the bad reputation of being sensitive to almost any kind of electrical disturbance. Based on the design aim to obtain reliability, the control system includes a line filter, regulated power supply, shielded signal leads, generous ground straps, component emission shields and the instruction that any inductive load wired to the control panel (e.g., solenoid coil) should be equipped with its own suppressor. Additionally, the entire control is packaged in a NEMA IV enclosure which adds an extra layer of shielding. The necessities and gains for each of these precautions is well documented in literature. With this required attention to detail, the control system has proven very reliable in all industrial environments to date.

CONCLUSIONS (ADAPTIVE COMPRESSOR CONTROL)

The general conclusions that can be reached from the experience of designing and utilizing a microprocessor based adaptive compressor control system are as follows:

1. Microprocessor based compressor control systems offer advantages in monitoring, diagnosing, communicating and control that as yet have undefined limits. The first generation control system discussed does more than is practical with standard type controls of the day, and stirs the imagination regarding what could be done in the future.
2. Efficient control of a plant air packaged compressor with its highly non-linear control loops has been attained using adaptive control and memory based learning techniques. These techniques of inherent machine intelligence allow the control to be built with only setpoint adjustments, since the PID characteristics are automatically and adaptably adjusted in internal digital logic.
3. Field experience with the adaptive compressor control has proven the reliability of the package to operate in varied environments and with varied power sources.

REFERENCES

1. Rabinowitz, M. D. and Hahn, E. J., "Stability of Squeeze-Film-Damper Supported Flexible Rotors", ASME, *Journal of Engineering for Power,* 99, Series A, 4, October, 1977, pp 545-551.
2. Feder, E., Bansal, P. N., Blanco, A., "Investigation of Squeeze Film Damper Forces Produced by Circular Centered Orbits", ASME, *Journal of Engineering for Power,* January, 1978, Volume 100, pp 15-21.
3. Bansal, P. N., Hibner, D. H., "Experimental and Analytical Investigation of Squeeze Film Bearing Damper Forces Induced by Offset Circular Whirl Orbits", ASME *Journal of Mechanical Design,* July 1978, Volume 100, pp 549-557.
4. Gunter, E. J., Barrett, L. E., Allaire, P. E., "Design of Nonlinear Squeeze Film Dampers for Aircraft Engines", ASME, *Journal of Lubrication Technology,* January 1977, pp. 57-64.
5. Bhat, S. T., Buono, D. F., Hibner, D. H., "Analysis of High Load Dampers", NASA #CR-165503, August 1981.
6. U.S. Patent Number 3,456,992, "Vibration Damping Device", April 7, 1967.
7. U.S. Patent Number 3,863,996, "Vibration Damper Bearing", February 4, 1975.
8. U.S. Patent Number 3,994,541, "Bearing Assembly", November 30, 1976.
9. U.S. Patent Number 4,097,094, "Journal Bearing Assembly with Flexible Support and Viscous Damping", June 27, 1978.
10. U.S. Patent Number 4,213,661, July 22, 1980.
11. Simandiri, S. and Hahn, E. J., "Effects of Pressurization on the Vibration Isolation Capacity of Squeeze Film Bearings", ASME, *Journal of Engineering for Industry,* 98, Series B, 1, February 1976, pp 109-117.
12. Choy, K. C. and Halloran, J. D., "Application of Hydrostatic Squeeze-Film Dampers", ASLE Transactions, Vol. 25, No. 2, pp 245-251, dated April 1982.
13. Li, D. F., Choy, K. C. and Allaire, P. E., "Stability and Transient Characteristics of Four Multilobe Journal Bearing Configurations", ASME, *Journal of Lubrication Technology,* 102-3, July 1980, pp 291-299.
14. Allaire, P. E., Li, D. F. and Choy, K. C., "Transient Unbalance Response of Four Multilobe Journal Bearings", ASME *Journal of Lubrication Technology,* 102-3, July 1980, pp 300-306.